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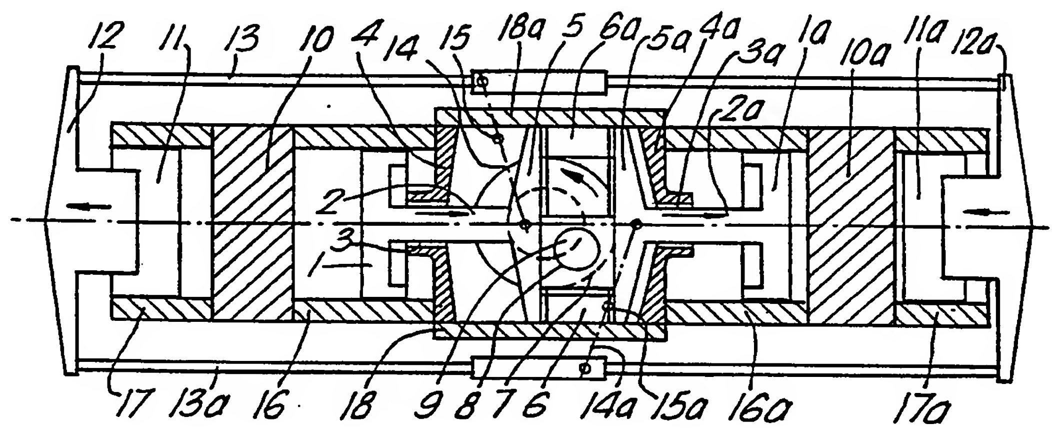

(57) Abstract

A double-acting two-stroke internal combustion engine comprises at least one pair, and preferably two parallel pairs, of opposed power units. Each aligned pair of the units includes a single counter-balanced crank throw therebetween carrying a bearing (8) for a slider (7). A pair of coaxial pistons (1), (11) and (1a), (11a) works in separate cylinders in each unit, the inner pistons (1), (1a) transmitting the drive through opposed members (5), (5a) of a Scotch yoke which straddles the slider (7) and the outer pistons (11), (11a) being mechanically linked together by rods (13), (13a) so as to reciprocate together. The two first pistons (1), (1a) are connected to the linkage between the two second pistons (11), (11a) by means of a system of first order levers (14), (14a) each having a stationary fulcrum (15), (15a) so that each of the first pistons (1), (1a) moves in the opposite direction to its respective second piston (11), (11a). Air/fuel mixture compressed by the second pistons (11), (11a) is directed through diffential pressure controlled non-return valves to primary ignition chambers communicating with offset combustion chambers formed partly in the first pistons (1), (1a) and their respective cylinders. The double-acting two-stroke internal combustion engine has many advantages including simplicity, good dynamic balance, low weight and a major economy of fuel consumption.

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INTERNAL COMBUSTION ENGINE

This invention relates to internal combustion engines and has as its object an integrated complete power unit which would have most or all of the virtues including the ability to meet any legislative requirements regarding pollution from carbon monoxide, oxides of nitrogen and hydro carbon emissions, and few of the drawbacks of known contemporary arrangements. This includes all forms of four-cycle and two-cycle engines together with any of the very wide range of transmissions and ancillary components.

Today, the four-stroke Otto cycle engine, although, for most of this century almost universally adopted as the standard, is still not without many problems and imperfections. These engines have been backed by astronomic sums of money and the inspirations of multitudes of experienced and highly qualified engineers. Despite this, most of the problems have still not been eradicated although much work is still going on.

Even now it is still the original problem of "those three idle strokes" which can be seen to be, in effect, the crux of the matter. With only one power stroke in any one cylinder every other crankshaft revolution, it is essential to have at least four cylinders in order to provide a relatively smooth power flow. Friction losses attend the transfer of expansion pressure into crankshaft rotation in the conventional engine and in similar fashion are experienced in the power return required for meeting the essential functional duties of compression, induction and exhaust

in the three other cylinders. Even at full power, this loss is about twenty percent with up to fifty percent at the lower road loads and one hundred percent at idle.

Long established practice usually has the four cylinders 'in-line' above a four throw crankshaft and, due to the firing order, those power impulses can occur back and forth along the line which thus causes many of the following problems -

(A) High frequency crankshaft torsional oscillations.

(B) Variations in quantity and quality of air/fuel mixture intake.

(C) Alternate tension and compression loads in connecting rods.

(D) Cyclic inertia loading variation at both dead centres.

(E) Secondary vibration at twice the fundamental frequency.

(F) Relatively high power/weight ratio.

(G) Up to fifty percent friction loss at the lower road load speeds.

(H) A somewhat limited operational speed range.

Engines operating on the two-stroke cycle have always been recognised as particularly efficient from the mechanical point of view, but do suffer from disadvantages arising from problems of scavenging and of introducing a fresh charge of mixture in the same stroke as the burnt gases are being exhausted.

The conventional crankcase compression, spark ignited two-stroke engine is not accepted as a worthy competitor to the almost universally accepted four-stroke engine, despite the drawbacks listed above. These views follow certain two-stroke failings such as:

a. Poor combustion due to some short circuiting of the fresh charge to the exhaust port, interface burning

from the conflict of in and out gas flows and the retention of a high percentage of the spent products of combustion with only a partial fresh charge.

b. Irritating, often staccato misfiring at lower loads.

c. Excessive fuel consumption.

These drawbacks, a to c above, were overcome by the invention described in my European patent no: 0,030,832 of which the essential ingredients are a pair of synchronized pumping pistons, one for the mixture and the other for air. Sandwiched between them, a cylinder head incorporates, on the mixture side, a differential pressure controlled non-return disc valve, then a primary ignition chamber into which a sparking plug provides ignition and finally, on the air side, a main combustion chamber. For the sake of operational clarity, the pumping actions were then described as using cranks and connecting rods in the usual way.

However, there remains:

d. Inadequate scavenging process allowing a degree of mixing.

e. Lubrication, up to 20 or more times that of a four-stroke engine, is grossly excessive. Much of this passes into the exhaust where it partially oxidises. This is a major source of legislation against such hydro carbon (HC) pollution.

According to the present invention a double-acting two-stroke internal combustion engine comprises: a pair of opposed power units driving a common output shaft located between the units and including a single counter-balanced crank throw carrying a slider bearing; each unit including a pair of co-axial pistons working in separate cylinders, both the first or inner pistons of the units transmitting the drive by through connection to the opposite members of a double slider crank chain

commonly known as a Scotch yoke which straddles the slider, and in which each inner piston inhales a volume of air for each cycle of operation, which volume of air is compressed beneath the inner piston against a crankcase wall towards the end of the power stroke and passed for scavenge through a port to the outer side of the inner piston in a direction perpendicular to the axis of that piston and the outer or second piston of each unit is synchronised with the first and serves to draw in a charge of mixture of air and fuel which, although still ignitable may be richer than stoichiometric, through an inlet valve or piston uncovered port, to compress it and force it through a differential pressure controlled non-return valve into a primary ignition chamber; the two outer or second pistons of the two units being mechanically linked together so as to move in the same direction as one another and to reciprocate together and each inner or first piston being connected to the linkage between the two second pistons by means of a system of first order levers each with a stationary fulcrum and is so located that the ratio of the two arms of the lever is the same as the ratio of the strokes of the first and second pistons, with the result that the first pistons move in the same direction as one another and reciprocate together, each moving in the opposite direction to its respective second piston; and the primary ignition chamber communicating with an offset combustion chamber formed partly in the cylinder head beyond the first piston partly by the cylinder wall and partly in the first piston, which combustion chamber is substantially an ellipsoid of revolution based upon a segment of a circle subtended by an angle of about 90°, the major axis of the ellipsoid being at right angles to the axis of the unit and having a radius similar to that of the bore of the cylinder of the first piston; whereby on each inward

stroke of the second piston a charge of mixture is compressed and forced through the differential pressure controlled non-return valve into the ignition chamber where it is ignited and passes through a passage directed substantially toward the centre of the ellipsoidal combustion chamber to the cylinder of the first piston to combine with air already compressed in the cylinder to provide the power stroke.

With this arrangement, all functional adverse loadings such as flow losses and compression will be deducted directly from the expansion power stroke and, piston ring friction only excepted, will include any piston skirt and bearing friction losses. It must be appreciated that in engines of the four-cycle multi-cylinder type, a number of friction losses attend the transfer of piston thrust resulting from the high pressure generated from combustion, into rotation of the crankshaft and that these same losses are experienced in the power extraction from the crankshaft for all the functional duties in all the other cylinders. Because in the two-stroke engine of this invention, the direct connection through the yoke between the pistons eliminates these piston skirt, connecting rod and crankshaft friction losses (not unlike an unnecessary middle-man of commerce), the mechanical efficiency is very high.

In order to retain this high mechanical efficiency, the slider/yoke mechanism is designed as fully self-aligning to avoid possible excess friction from the slightest malalignment, deformation or deflection in the unit. Obviously this is of the greatest value and importance in a high-speed, lightweight quantity production engine.

In an engine intended for use as automotive propulsion, most of the running will be at speeds well below eighty percent of maximum and at loadings rarely

exceeding fifty percent. In overall use, except in the case of heavy long distance goods vehicles, it might be unusual to average more than a twenty five percent loading. Consequently it is the amount of low load running which governs fuel consumption. Under these conditions, the normal form of four-cycle engine will be well throttled down which results in a partial charge, a fair percentage of unscavenged spent gas and a low effective compression ratio while still suffering a substantial friction loss.

Due to operation on different principles, the engine of this invention will not experience the above problems. First, at a cruising speed of about fifty percent maximum, only the mixture pumps (usually thirty five percent of the total engine displacement) are throttled to about forty percent while the remaining sixty five percent of air intake is unthrottled. Thus the compression ratio pertaining to this condition would be $0.4 \times 0.35 + 0.65 = 79\%$ of maximum which would ensure approaching double the compression ratio efficiency. Consequently as, obviously, a lesser quantity of heat (air/fuel mixture) is added to the compressed air, the ratio of pressure multiplication and temperature following ignition will be substantially reduced.

Further, in the work per cycle equation, the term involving the compression ratio, $r^{1.3}-r/r-1$, the useful work per cycle increases in strict proportion to the value of the term. In the two cases noted above, those values are about 0.58 for the four-stroke and 1.02 for the engine of this invention. From this then, the reduction of heat added multiplied by the compression ratio term would indicate that the work per cycle to be 0.44 for the two-cycle compared with 0.86 for the conventional engine. As however, the two-cycle has twice as many power strokes this shows that as 0.88 : 0.86,

this gives approximately the same output of power for one revolution of the engine crankshaft.

An engine according to the invention also leads to the following advantages:

a) Lower maximum to mean pressure ratios results in less weight all around and an increased mechanical efficiency.

b) Complete combustion of less fuel in the same quantity of air raises thermal efficiency toward the air cycle efficiency.

c) Heat added at the higher temperature will not produce the same pressure and temperature increase as when added at lower temperature.

d) Lower mean pressures ensures less inflation of piston rings and this reduces the ring friction.

e) The higher the percentage of added heat which increases the thermal efficiency, the greater the engine reliability, as waste heat flow causes most engine ill effects.

f) In the absence of piston side thrust, only a minute quantity of oil, sufficient only to keep the piston rings in good condition, passes into the working or air pumping cylinders.

g) This insignificant passage of oil (about 0.05% of normal) into the exhaust tract, virtually eliminates this HC form of pollution.

h) At least 50% better scavenge than obtains with conventional two-cycle engines.

In order to ensure the improved scavenging the portion of each first piston which surrounds the respective combustion chamber provides about half of the total surface area. This deflector is so shaped as both to separate the air from the exhaust and also cause directed action of the transfer entry air to follow behind the combustion gas leading to improved scavenging.

The invention will now be described in more detail, with reference to the accompanying drawings in which:-

Figure 1 is a sectional elevation;

Figure 1a is a diagrammatic view similar to Figure 1;

Figure 2 is a plan view of a double unit with half in section;

Figure 2a is an enlarged view in particular to show the shape of the deflector and the throat through which the scavenge air is funnelled;

Figures 3 and 3a are comparative crank motion curves for a four cylinder four-cycle engine and a four cylinder two-cycle engine in accordance with the invention;

Figure 4 is a diagram of all the forces during one stroke only of an axial pair as shown in Figures 1 and 2 as calculated by conventional formulae;

Figure 5 shows one form of actuating lever;

Figure 6 is the arrangement of the ignition/starter switch;

Figure 7 illustrates a transverse vertical section of the cylinder heads through the primary ignition chamber; and

Figure 8 is a central vertical transverse diagram corresponding to the plan view of Figure 2.

The engine illustrated in Figures 1 and 2 includes two pairs of axial units arranged side by side, each unit of each pair including a pair of pistons as best seen in Figure 1. An inner pair of main air pistons 1 and 1a are attached to tubular piston rods 2 and 2a passing through bearing bushes 3 and 3a which are mounted in crankcase end walls 4 and 4a. Within the crankcase proper, the tubular piston rods 2, 2a are attached to

members 5,5a forming 'T' shapes. The ends of these members are fastened to bridge pieces 6,6a thus enclosing a rectangular opening into which a slider 7 is free to traverse between the concave faces of 5,5a of the opening.

In the parts 5, 5a, 6 and 6a assembled as a yoke, the concave faces of the main members 5 and 5a, form cylindrical bearing faces perpendicular to the axis of the unit proper. Slider 7 is the longitudinal central third of a cylinder whose axis is likewise perpendicular to the unit axis. The bearing faces of the slider are, with working clearance, in contact with those of the members 5 and 5a; both being machined to the same radius (plus working clearance) as the slider cylinder.

In addition the bridge parts 6 and 6a, to which parts 5 and 5a are attached to form the complete yoke, also have their outer bearing faces mated to bearing faces on the crankcase caps 18 and 18a. These faces are machined to radius equal to their distance from the unit axis plus working clearance as required.

The result of this is that the slider 7 has a degree of freedom about its perpendicular axis while still maintaining full bearing contact with the concave faces 5 and 5a. Equally so has the yoke assembly the same freedom about the unit axis and with full bearing contact.

An anti-friction bearing 8 is mounted within the slider and bears on crankpin 9 of the counter-balanced crankshaft. Members 5,5a,6,6a and 7 together constitute a self-aligning form of a Scotch yoke and with constant rotational angular velocity of 9 will endow the whole sub-assembly of 1,1a, 2,2a, 5,5a, 6 and 6a with pure harmonic motion. With the exception of parts 3,3a, 4 and 4a, the static members, the parts so far described form the air pumping sub-assembly mechanism.

The preferred crankpin bearing 8 in this engine is of the low friction needle roller type because these operate at a constant angular velocity and have extremely modest lubrication needs. Due to the overall arrangement, the bearing has to withstand loadings which rarely exceed but fifteen percent of that which is customary.

Cylinder heads 10,10a also static and again in axial alignment are suitably positioned outboard of the two pistons 1,1a. The mixture pistons 11,11a with members 12,12a attached and perpendicular to the general axis are connected at their outer extremities by rods 13,13a guided by bearings 113. This mixture sub-assembly 11,11a, 12,12a, 13 and 13a is caused to reciprocate axially in the opposite direction to the air pumping sub-assembly by means of a pair of asymmetrical first order actuating levers 14,14a pivoted about static fulcrum bearings and connected to the rods 13,13a in the region of the bearings 113. The operating length of the two arms of these levers is in the same proportion as the design stroke lengths of the air and mixture pistons. Further, with equality of the product of stroke times weight of the two sub-assemblies, operation of the engine will be in dynamic balance and devoid of vibration.

This mechanism, while usefully compact, has the added advantage to the engine when working, of simple harmonic motion. Thus, not only do both cylinders have identical crank angle/displacement ratios, but the inertia values are the same at both ends of the stroke. Further the slower rate of pressure rise per degree of crank rotation (as compared with the conventional connecting-rod/crank arrangement) aids the attainment of complete combustion.

As shown in Figure 1, in order to maintain a reasonable degree of clarity, the actuating levers 14,14a

are shown as dashed lines and the fulcrum bearings are indicated at 15,15a. Although the placement of the levers may be as shown in the diagram, in practice they may be in other operating positions and may make use of links. The drawings are largely diagrammatic and omit accepted detail such as fastenings, seals, piston rings and cavities for cooling liquid. Further, for the same reason, the static parts 3,3a, 4,4a, 10 and 10a together with the air cylinders 16,16a, mixture cylinders 17,17a and crankcase caps 18,18a are shown cross hatched. Thus two cylinders each with both air and mixture pistons, the yoke, crank, levers and link gear, all as listed above plus only the crankcase side walls (19,19a to be found in Figure 2) form one complete pair of units or axial group.

The entire process of combustion and the values of expansion are unlike the characteristics of either the four-stroke or two-stroke cycles of convention or even their multi-carburettor or fuel injection forms which at least provide superior distribution.

In a complete power unit, as shown in the plan view of Figure 2, two of the twin cylinder axial groups as outlined earlier are arranged one on either side of a modified hydraulic converter, coupling or clutch 20. Each of the fully counter-balanced single throw crankshafts 21 shown (21a not shown) are taper fitted and screw fastened into opposite sides of the hub of the central unit and at 90° to each other. The total crankshaft weight will be under ten percent of that of a conventional engine of similar power and yet it will be completely devoid of any high frequency torsional oscillations and the complete engine has no need either for heavy thick rubber engine mountings or any form of vibration damper. Thus with this arrangement, there will be four power strokes per revolution giving an exceptionally smooth

torque as may be seen in Figures 3 and 3a in comparison with a four-cycle engine.

There is a striking difference between the two crank motion curves as shown in Figures 3 and 3a which relate to two four cylinder engines of about the same power output, Figure 3 showing a four-cycle engine and Figure 3a showing a two-cycle engine in accordance with the invention. The curves depict the condition without a torque smoothing flywheel. Obviously the one engine with torque extremes of about two hundred and forty nine percent above and below the mean and, in particular, with forty four percent of its running time well below negative (shown hatched), would require quite a heavy flywheel even for idling at many hundreds of revolutions per minute. This compares most unfavourably with the two-cycle engine having variations only fifty two percent above and below the mean.

Returning to Figure 2 with particular reference to the matter of balance and smooth running as well as economy, there are two single carburettors 22 (the other not shown), one for each in-line pair of cylinders. Each carburettor is mounted on a simple tubular manifold 23 (the other not shown). With this the air/fuel charge will be uniform in both quantity and quality in each of the cylinders as the intake conditions are identical; this is almost impossible to achieve in any multi-cylinder four-cycle engine.

Arrows are drawn on both Figures 1 and 2 to show the directions of movement and of the crankshaft rotation. With anti-clockwise rotation of the crankshaft the illustrated positions of the moving parts and the stages of operation are as follows:-

i. Air piston 1a is approaching cylinder head 10a and is near the end of the compression stroke.

ii. Air is being drawn into the space between 1a and 4a through the air inlet 24a.

iii. Mixture is being compressed close to opening the valve 28a by movement of piston 11a.

iv. Piston 1 is reaching the moment of uncovering the exhaust port 27 toward the completion of the expansion stroke.

v. The previously inhaled air between piston 1 and wall 4 is being compressed.

vi. Mixture piston 11 is inhaling air/fuel mixture from the carburettor 22 through manifold 23 port 25 and valve 26.

About 35° later -

<u>iii.</u> Piston 11a will have raised the mixture pressure to the maximum, overcome the bias pressure of disc valve 28a, flowed into primary ignition chamber 29a, been ignited by sparking plug 30a and spurted into the compressed air in the main combustion chamber 31a, there meeting an excess of oxygen for complete combustion.

<u>iv</u>. Following combustion in the other cylinder 16, the centre of gravity of the burning gas flows strongly away from the main chamber 31, maintaining high inertia direct to the exhaust port 27. Immediately after the exhaust port opens and the strong blast exit of the spent gas, the transfer port 32 opens. This releases the pressure scavenge air which is then directed around the curl of the deflector 33 through a throat formed between the tip of the deflector and the cylinder wall. This keeps the scavenge air clear of the exhaust flow and directs it up the cylinder wall toward the main combustion area 31 and expanded to fill in behind the exhaust expulsion, thus resulting in a high degree of scavenge which approaches that of the four-cycle engine.

Figure 4 shows (as calculated by conventional formulae) all the forces during one stroke only of an axial pair as shown in Figures 1 and 2. The dash/dot line 36 is of the expansion curve of pressure in the one cylinder while the dashed line 37 which starts at maximum negative, crosses the zero line 39 at the 90° ordinate and continues the sine wave to maximum positive represents the total inertia of the reciprocating parts. The dotted curve 40 is that of the compression pressure in the opposing cylinder as this is directly yoked to the subject cylinder and must be considered as energy subtracted from the total energy generated. Finally the solid line 38 represents the resultant of the values of 36, 37 and 40. Another identical curve would follow this one and would represent the crank moment over the one complete revolution. In this case there would, of course, be zero torque at certain points, however, combined with an identical curve displaced 90°; the four cylinder crank moment would be as depicted as Figure 3a.

Figure 5 shows one form of the actuating lever 14,14a in detail in place of the dashed lines of Figure 1. The rollers at 41 and 42 have needle roller bearings on hardened pins. The levers are in pairs one on each side of both yoke members 5, 5a with the ends at 42 on either side of the push/pull rods 13,13a. This makes a total of four levers on each axial group and is required for complete balance and safety to handle the calculated loadings. Being close fitted to the yoke members, the levers operate between the yoke and the circular discs of the crankshaft throws. The gaps needed may be seen in the Figure 2 cross section at CC on Figure 2.

With this general engine arrangement, there are a number of factors which make engine starting efficient, quiet and reliable while dispensing with a large heavy battery and the customary starter motor with its gearing

which, for about ninety eight percent of its life, is just a passenger.

Due to the absence of 'stiction' from piston skirt and the multiple plain bearing cold oil films, the engine rotates very easily. Because of this, the engine will make a number of turns when switched off after running. This ensures that the primary ignition chamber and adjacent passages remain charged with ignitable air/fuel mixture. Under these conditions, it is almost inevitable that, when brought to rest, all four main pistons will stop about half way between the normal 90° firing points.

Due to this, a spark in the primary ignition chamber will start the engine turning for part of the stroke. Now, with the normal ignition switched on, the primary winding of the coil would be saturated, so that any momentary interruption (break) in the circuit would cause collapse of the magnetic field with the very high rise in the secondary voltage and the required spark. Now if the jump spark electrode on the distributor rotor arm had an elongated arc, this arc would by-pass the normal electronic or mechanical make/break arrangement and not interfere with the correctly organised next spark igniting the charge in the following cylinder to top dead centre some 50° later. The engine should then be running normally. If not, pressing the push button above the key will then bring in the hydraulic positive starting system.

Engine starting in detail relies upon two switching devices, one of which is a microswitch 55 (Figure 6) with a nylon roller operator 56. The key cylinder is integral with cam plate 57. This plate has a spring loaded cam 58 and a normal running cam 59 as well as the customary detents to retain the operating positions. On clockwise key rotation cam 58 first

depresses the roller which operates the switch to close the circuit between leads 60 and 61 to saturate the ignition coil. Instantly after this (at the normal speed of key operation) the roller 56 drops off cam 58 which opens the circuit. This is the make/break which causes the starting spark earlier mentioned. Immediately after this, the plate 57 comes to rest with cam 59 causing the circuit to be closed for continuous running. In switching 'off', the key rotates anti-clockwise, the roller leaves cam 59, the circuit opens and spring loaded cam 58 is pushed away from the operating position. This prevents a further make/break which would cause a 'wild spark' to disturb the normal stopping procedure.

Figure 7 shows a transverse section of the cylinder heads 10, 10a through the centre of the primary ignition chambers 29,29a and the axes of the sparking plus 30,30a. It will be seen that the machining and screw threading are identical both above and below each of the primary ignition chambers 29,29a. This enables dual ignition to be provided when required as in aircraft.

While the system as described above provides, in most cases, starting of fair reliability, there may occur some failures especially following any longer periods of non-use. Should such a failure to start occur, there is an auxiliary means linked to the transmission and/or the ancillary system as may be seen later.

The integration of the power unit extends now to the ratio change form of transmission. This consists of two sets of silent chain drives one on either side of either a fluid coupling, clutch or a hydrodynamic converter. The basic arrangement is much the same in each case but the end product is to meet the rather different terms of reference dictated by the intended

service required, the economic conditions and factors of purchaser/user preference. The basic arrangement is shown on Figure 6 which is a central vertical transverse diagram as seen from the right of the plan part section in Figure 2. The two axial cylinder groups are indicated as A and B. The two silent chain sets are the same but reversed; each has a smaller sprocket of say twenty nine teeth and a larger one of say forty one teeth and both have the same number of links in the chain which should always be an even number so that the centre distance of both chain sets is the same.

More specifically and turning to the fluid coupling or other arrangement, on the A or low gear side, the driver 43 of twenty nine teeth is fixed to the output or runner of the coupling 20. The chain 44 and the driven wheel 45 of forty one teeth forms a reduction gear of 1.4138:1 to the final shaft 46 by means 47 of a two-way synchronising positive clutch mechanism either direct to 46 or to a one-way clutch 48 to the shaft 46.

On the B side, the driver sprocket 34 (also shown in Figure 2) of forty one teeth is fixed to the impeller of coupling or clutch primary member 20 and, through chain 49, drives the driven sprocket or chain wheel 50 of twenty nine teeth which, in turn, through an oil pressure operated friction clutch 51 rotates the drive shaft 46. The oil pump in casing 52 is also driven by the sprocket 50 even without it being clutched to the final shaft 46. Here, on this B or high gear side, the speed ratio is 0.7073:1 the reciprocal of the A chain set and providing a total range of 1.9988:1 which, with the operational characteristics of the engine, will be completely capable of handling all normal use.

The number of teeth on the sprockets should, for preference, always be prime numbers and the number of chain links should be even but never any multiples of the

sprocket numbers. This 'hunting-tooth' principle ensures perfectly even, although negligible, wear because every link in the silent chain contacts, in sequence, every tooth on each sprocket.

The commercial silent chains used have a specific form of construction providing an exceptionally high mechanical efficiency which virtually eliminates the velocity change which generates a chordal action in conventional chains. Further, this form of silent chain engages both sprockets with less sliding action and virtually no entering impact. These factors eliminate the inevitable 'thrash' noise and also the very high frequency vibration which, together with the absence of uneven wear because of the 'hunting tooth' design, will ensure an exceptionally smooth drive of unusually high mechanical efficiency. In practice the second stage group comprising parts 45,46,47,48,50,51,52 and 53 which is the planetary cluster to be described later, are not as is shown in Figure 8. They will be positioned either to the rear or forward of part 20 as will be dictated by the overall design of the vehicle.

Dealing with the auxiliary starting means, here, the second switch receives current from the common line noted above, is push button operated and is normally in the open circuit position. Now, as earlier had been mentioned, sprocket 50 was directly connected to an oil pump contained with casing 52 and is the source from which a supply of pressure oil is available mostly for operation of the transmission. Normally the pumped oil is stored in a spherical diaphragm type accumulator and under control to 'cut-in' at about 15 bar and 'unload' the pump at about 20 bar.

In operation, current from this second switch will energise a solenoid connected to a selector valve which changes over the hydraulic lines to and from the

pump and at the final movement, will open the non-return valve to the accumulator. The high released pressure oil then flows direct to the normal suction side of the pump 51 which, in the normal way with such pumps, now functions as a motor and by being connected direct to the sprocket 50 of 29 teeth, turns sprocket 34 of 41 teeth by means of the chain 49 at a torque increase of 141%. Operation of the solenoid/selector cuts out the 'unloading' hydraulic circuit at starting and restores it at idle; the 'cut-in' only operates at opening throttle.

Item 53 is a compound planetary/clutch of pressure oil operation. In general it is of one of the well known 'text-book' designs and would be provided to give a lower ratio, neutral and reverse in particular when in combination with either the fluid coupling or a convertor.

As, in any case, a two-cycle engine will run equally well in either direction of rotation, it is clear that, if a spark be arranged to start rotation in a cylinder after top dead centre as has been described, it must follow that, if such a spark is arranged to occur before top dead centre, the engine will start and run in the reverse direction. This ability would be available should the engine be used for a reduction-gear marine installation. In such a case parts 34, 47, 48, 49, 50, 51 and 53 would not be needed, shaft 46 could be the propellor shaft, part 20 would be a normal friction disc clutch to provide neutral and the oil pump 52 could still be fitted. The cam plate 57 would have a mirror relationship extension bearing similar cams 58a and 59a as shown in Figure 6. In addition the plate 57 would carry a link to the ignition distributor to provide its rotation to a suitable reverse setting.

The number of teeth on the pairs of sprockets as quoted (29 and 41) are not fixed for all applications.

These given are quoted more or less as the preferred limits for load and life. Particularly in the case of a convertor as part 20 where there would be a torque magnification of about 2 : 1, the two chain sets could have a much closer ratio by using sprockets of 31 or 37 teeth.

	TEETH	RATIO	RATIO^2
a	29:41	1·4138	1·9988
b	31:41	1·3226	1·7492
c	29:37	1·2759	1·6278
d	31:37	1·1935	1·4246
e	37:41	1·1081	1·2279

These two chain ratios, whether wide (a and b) or close (d and e) will be then most used. The high ratio B side drive will overrun the A side due to the one-way clutch 48. Additionally, on slowing, the lower ratio will take over without jerk.

In summary, this specification discloses an invention relating to a totally integrated internal combustion engine power unit including the transmission and starting system. The paramount aim of this design is efficiency and economy from a really simple concept. This should not be considered as just a reduction in the number of parts but far more in the lessening of complex and/or intricate machining operations without the sacrifice of the highest quality and efficiency of function. Accuracy of alignment and efficient assembly is assured by the use of four through bolts 100 in Figure 7. Despite this simplicity, the elimination of torsional oscillation and secondary vibration with complete dynamic balance is ensured without need of any additional components. Further, only a minimal flywheel is needed to smooth the very small torque variation even at an idle speed which is far lower than is customary.

Finally, the substantial reduction of friction loss, the stratified combustion process and lower weight of the complete unit, will result in a major economy of fuel consumption.

CLAIMS

1. A double-acting two-stroke internal combustion engine comprising: a pair of opposed power units driving a common output shaft located between the units and including a single counter-balanced crank throw carrying a slider bearing; each unit including a pair of co-axial pistons working in separate cylinders, both the first or inner pistons of the units transmitting the drive by through connection to the opposite members of a double slider crank chain commonly known as a Scotch yoke which straddles the slider, and in which each inner piston inhales a volume of air for each cycle of operation, which volume of air is compressed beneath the inner piston against a crankcase wall towards the end of the power stroke and passed for scavenge through a port to the outer side of the inner piston in a direction perpendicular to the axis of that piston and the outer or second piston of each unit is synchronised with the first and serves to draw in a charge of mixture of air and fuel which, although still ignitable may be richer than stoichiometric, through an inlet valve or piston uncovered port, to compress it and force it through a differential pressure controlled non-return valve into a primary ignition chamber; the two outer or second pistons of the two units being mechanically linked together so as to move in the same direction as one another and to reciprocate together and each inner or first piston being connected to the linkage between the two second pistons by means of a system of first order levers each with a stationary fulcrum and is so located that the ratio of the two arms of the lever is the same as the ratio of the strokes of the first and second pistons, with the result that the first pistons move in the same direction as one another and reciprocate together, each moving in the

opposite direction to its respective second piston; and the primary ignition chamber communicating with an offset combustion chamber formed partly in the cylinder head beyond the first piston partly by the cylinder wall and partly in the first piston, which combustion chamber is substantially an ellipsoid of revolution based upon a segment of a circle subtended by an angle of about 90°, the major axis of the ellipsoid being at right angles to the axis of the unit and having a radius similar to that of the bore of the cylinder of the first piston; whereby on each inward stroke of the second piston a charge of mixture is compressed and forced through the differential pressure controlled non-return valve into the ignition chamber where it is ignited and passes through a passage directed substantially toward the centre of the ellipsoidal combustion chamber to the cylinder of the first piston to combine with air already compressed in the cylinder to provide the power stroke.

2. An internal combustion engine according to claim 1, in which the faces of the members of the Scotch yoke which engage the slider are concave and mate with corresponding convex faces on the slider.

3. An internal combustion engine according to claim 1 or claim 2, in which the faces of the members of the Scotch yoke which engage the crankcase are convex and mate with corresponding concave faces on the crankcase.

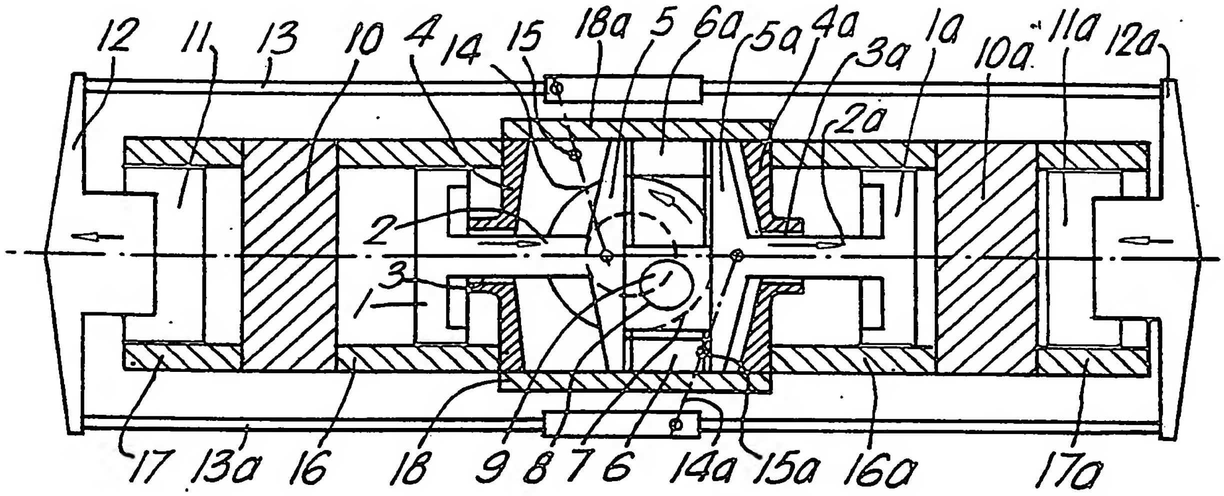
4. An internal combustion engine according to any one of the preceding claims, in which the part of each first piston which partly surrounds the respective combustion chamber acts as a deflector to cause swirling action of the transfer entry air to follow behind the combustion gas leading to improved scavenging.

5. An internal combustion engine according to any one of the preceding claims, in which each of the primary

ignition chambers is capable of being associated with two sparking plugs for providing dual ignition.

6. An internal combustion engine according to any one of the preceding claims, in which there is a total of four of the power units arranged as two parallel pairs on either side of part of a transmission, each of the counter-balanced single throw crankshafts being taper fitted and fastened into opposite sides of the central transmission part and at 90° to each other.

Fig.1.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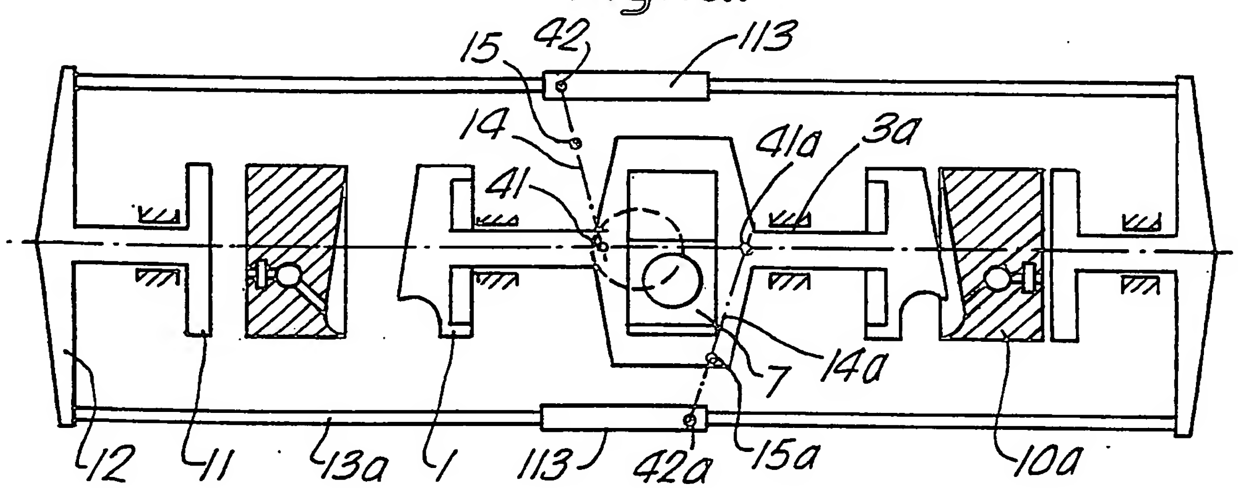
17a

Fig.1a.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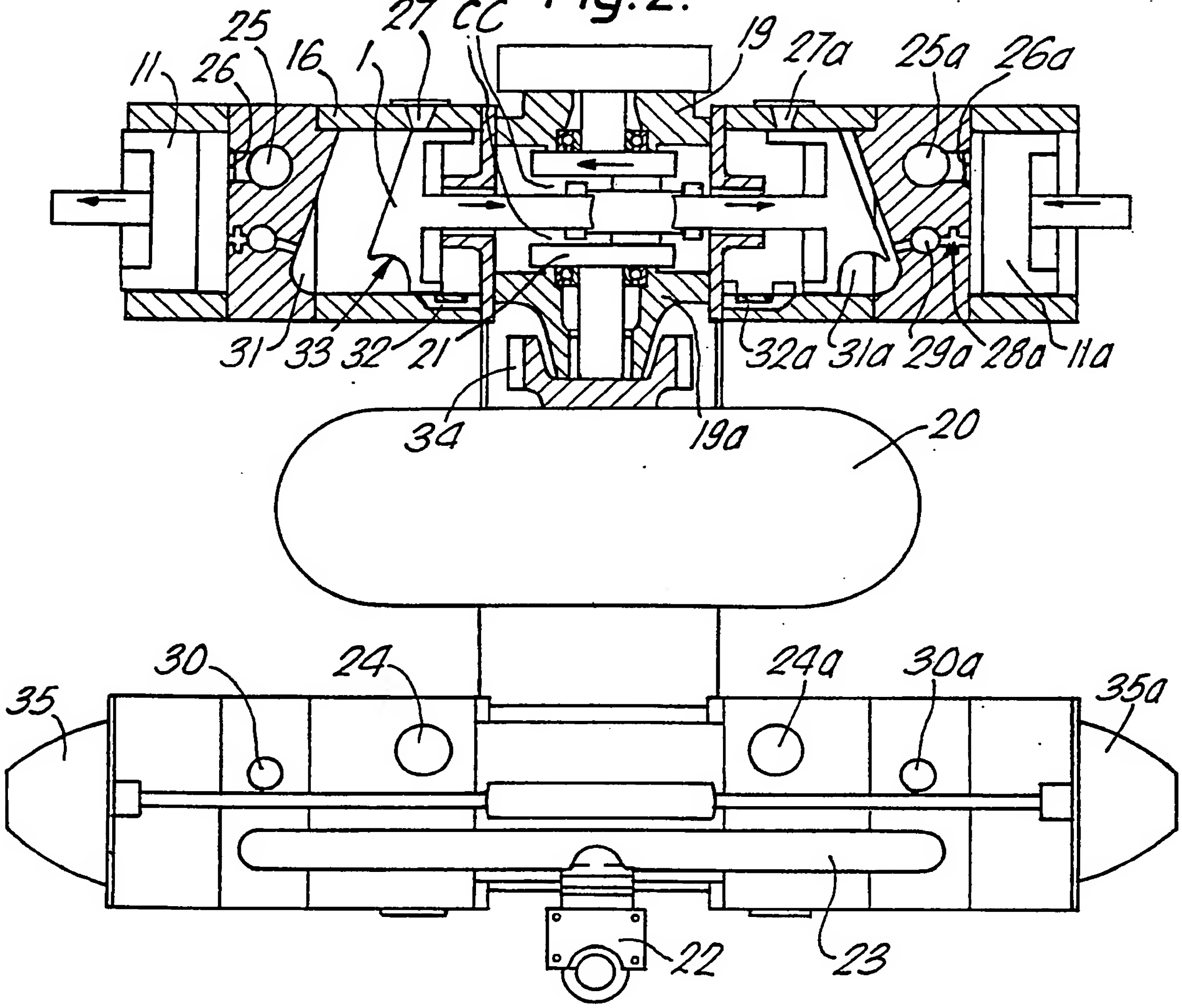
15a
10a

Fig. 2.



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25a
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35a
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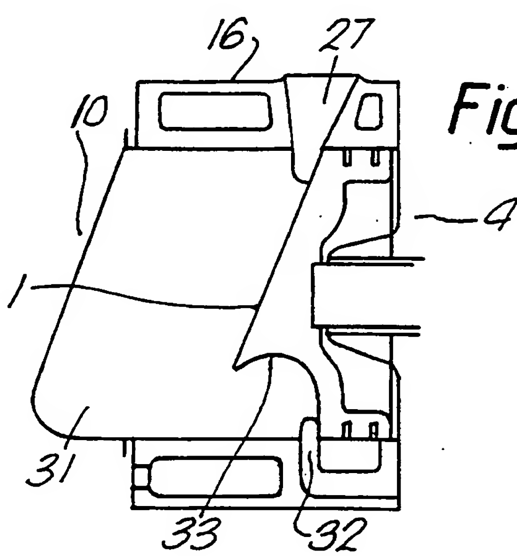


16
27
10
4
1
31
33
32

Fig. 2a.

Fig.3.

FOUR CYLINDER
4 CYCLE

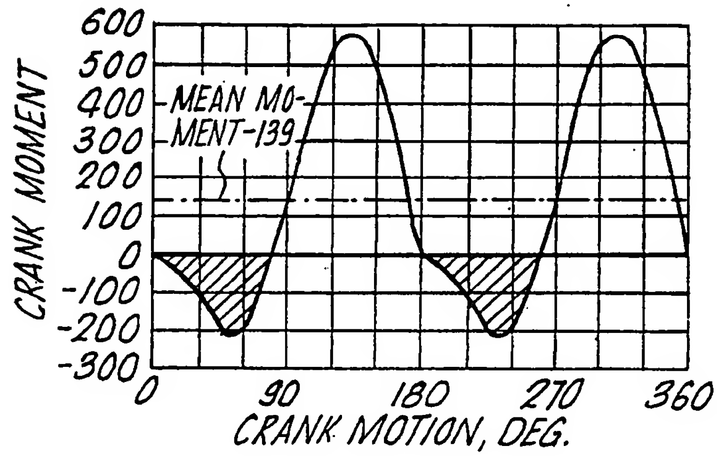
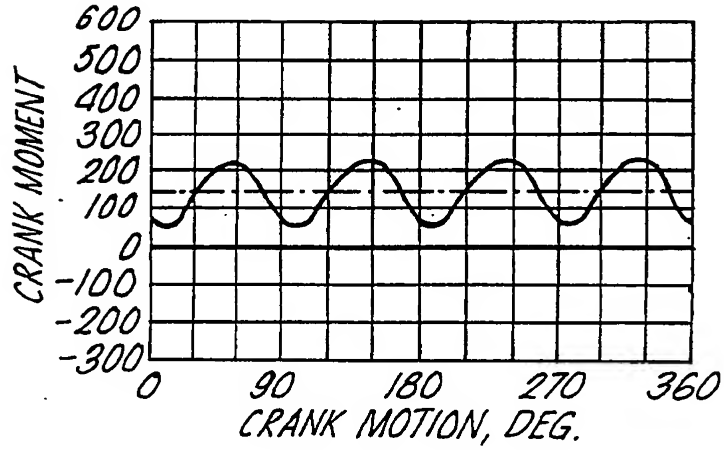


Fig.3a.

FOUR CYLINDER
2 CYCLE



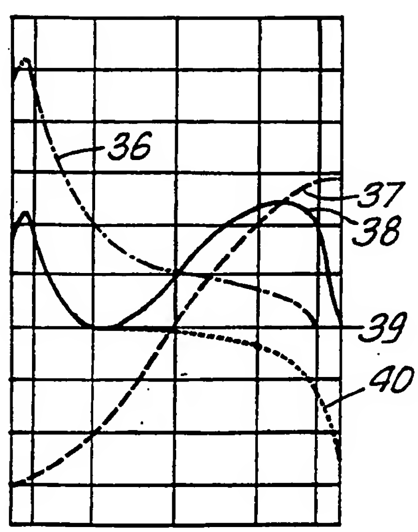


Fig.4.

DIAGRAM OF FORCES
IN ONE CYLINDER

Fig.5.

ONE FORM OF THE ASYMETRICAL
ACTUATING LEVER WITH ROLLERS

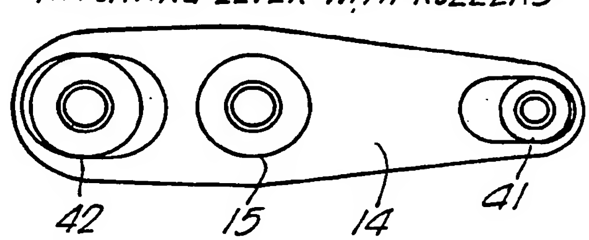


Fig.6.

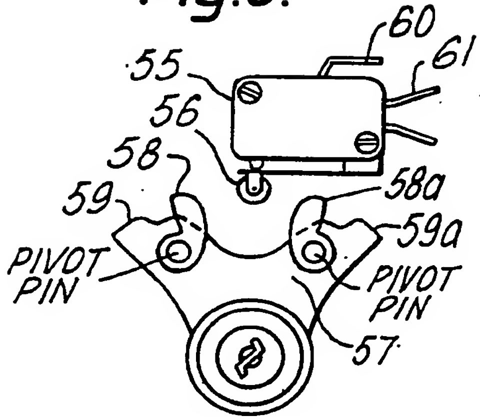


Fig.7.

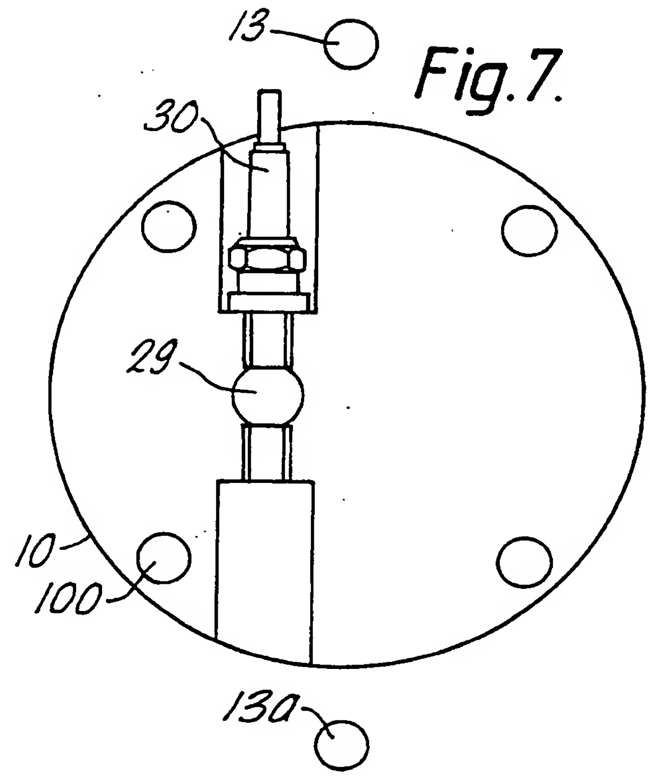
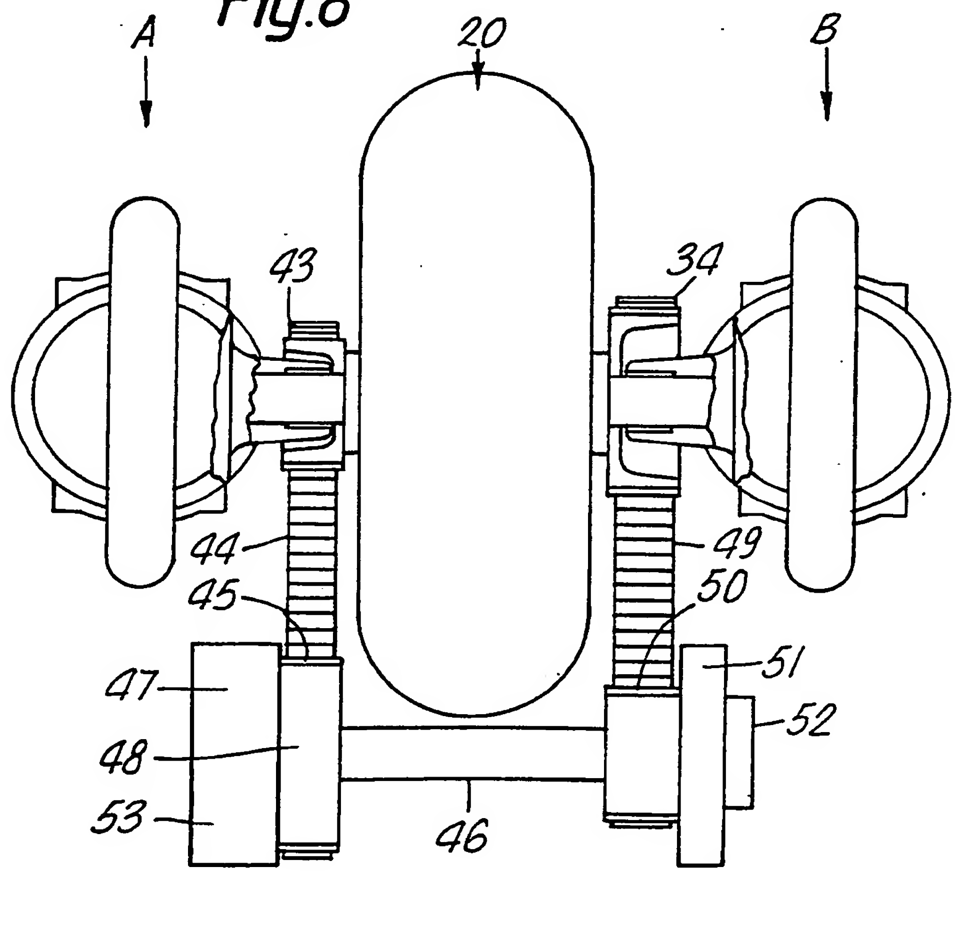


Fig.8



INTERNATIONAL SEARCH REPORT

International Application No PCT/GB 89/00123

I. CLASSIFICATION OF SUBJECT MATTER (if several classification symbols apply, indicate all) [6]

According to International Patent Classification (IPC) or to both National Classification and IPC

IPC^4: F 02 B 75/28; F 01 B 9/02

II. FIELDS SEARCHED

Minimum Documentation Searched [7]

Classification System	Classification Symbols
IPC^4	F 02 B; F 01 B

Documentation Searched other than Minimum Documentation to the Extent that such Documents are Included in the Fields Searched [8]

III. DOCUMENTS CONSIDERED TO BE RELEVANT [9]

Category [10]	Citation of Document, [11] with indication, where appropriate, of the relevant passages [12]	Relevant to Claim No. [13]
A	US, A, 4559838 (NEUENSCHWANDER) 24 December 1985, see figure 1; column 2, lines 16-61	1
A	US, A, 4516539 (ANDREEN) 14 May 1985, see column 3, line 34 - column 4, line 60	1
A	FR, A, 2526862 (FICHT) 18 November 1983, see figures 2,4; page 6, lines 7-35	1,2,3
A	GB, A, 1015189 (LINDSAY) 31 December 1965, see figures 10,11; page 4, lines 12-74	1,4,5

° Special categories of cited documents: [10]

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"T" later document published after the international filing date or priority date and not in conflict with the application but cited to understand the principle or theory underlying the invention

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"Y" document of particular relevance; the claimed invention cannot be considered to involve an inventive step when the document is combined with one or more other such documents, such combination being obvious to a person skilled in the art.

"&" document member of the same patent family

IV. CERTIFICATION

Date of the Actual Completion of the International Search	Date of Mailing of this International Search Report
20th April 1989	18 MAY 1989
International Searching Authority	**Signature of Authorized Officer**
EUROPEAN PATENT OFFICE	P.C.G. VAN DER PUTTEN

ANNEX TO THE INTERNATIONAL SEARCH REPORT ON INTERNATIONAL PATENT APPLICATION NO.

GB 8900123
SA 26806

This annex lists the patent family members relating to the patent documents cited in the above-mentioned international search report. The members are as contained in the European Patent Office EDP file on 12/05/89
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Patent document cited in search report	Publication date	Patent family member(s)	Publication date
US-A- 4559838	24-12-85	None	
US-A- 4516539	14-05-85	CA-A- 1221311	05-05-87
FR-A- 2526862	18-11-83	DE-A- 3218311 GB-A,B 2123495 JP-A- 59037230 US-A- 4512290	24-11-83 01-02-84 29-02-84 23-04-85
GB-A- 1015189		None	

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For more details about this annex : see Official Journal of the European Patent Office, No. 12/82